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ACOUSTIC SCALING: A re-evaluation of the acoustic model of Manchester, Studio 7

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Summary

The reasons for the reconstruction and re-evaluation of the acoustic scale model of a large music studio are discussed. The design and construction of the model using mechanical and structural considerations rather than purely acoustic absorption criteria is described and the results obtained are given. The results confirm that structural elements within the studio gave rise to unexpected and unwanted low-frequency acoustic absorption. The results also show that, at least for the relatively well understood mechanisms of sound energy absorption, physical modelling of the structural and internal components gives an acoustically accurate scale model, within the usual tolerances of acoustic design. The poor reliability of measurements of acoustic absorption coefficients, especially when the room already contains significant amounts of other acoustic absorption is well illustrated.

The conclusion is reached that such acoustic scale modelling is a valid and, for large scale projects, financially justifiable technique for predicting fundamental acoustic effects. It is not appropriate for the prediction of fine details because such small details are unlikely to be reproduced exactly at a different size without extensive measurements of the material's performance at both scales. A combination of the two methods, namely physical modelling of materials whose acoustic behaviour is unknown and absorption coefficient modelling of materials whose performance is known, will result in an acoustic model sufficiently accurate to enable serious mistakes to be avoided.

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ACOUSTIC SCALING: A RE-EVALUATION OF THE ACOUSTIC MODEL OF MANCHESTER, STUDIO 7

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1. Introduction to acoustic scale modelling

1.1. General

In the fields of building and architecture, the concept of constructing a small scale model of a proposed space for the purpose of making measurements and assessments is self-evident. It has been used in the past by architects and designers to enable them to plan the space effectively, and, more recently, by ventilation engineers to check air distribution systems. Such modelling becomes more cost effective as the size of the proposed construction increases and the modelling costs become a smaller part of the total. It also helps avoid mistakes which, on a large scale, can be very expensive to rectify.

These considerations apply equally well to the modelling of the acoustics of the proposed construction, assuming that the acoustic properties of the interior are of any importance. It is therefore not surprising that acoustic scale modelling techniques, of a more or less primitive form, have been applied for many years. The earliest applications used two-dimensional models of sections through the proposed building[1] together with ripple-tanks[2], in which the waves on the surface of a shallow liquid can represent the sound waves, and thus depict the acoustic behaviour of the space. Using such methods, especially in connection with the design of concert halls, enables the distribution of the sound energy, for example, beneath balconies to be studied and gross acoustic defects such as focussing of the sound energy to be avoided. Prior to the development of acoustic measurement techniques and criteria for acoustic quality, this two-dimensional approach represented the useful limit of acoustic scale modelling methods.

With the development of transducers and measurement techniques it became possible to utilize three-dimensional models using three-dimensional acoustic waves in a compressible fluid medium to represent the passage of the sound energy through the volume of the proposed space[3]. Air is most commonly used for this purpose although other gases may be used, for example, nitrogen (see Section 1.2). To maintain the proper scale for the model sound waves, the wavelength must be reduced by the scaling factor which, in air, is equal to the geometric scale factor of the model. For other fluids the ratio of sound velocities must also be included. In general, for modelling large structures on a reasonable scale, this scaling of the wavelength is equivalent to an increase in the frequency of the sound.

As an extension of the two-dimensional ripple-tank models, three-dimensional geometric models can be used in the same way to study reflections, energy distribution and focussing. Using air, scale factors up to 50:1, or even 100:1 are technically feasible[4,5] if a very limited frequency range is acceptable. At a scale factor of 50:1 in air, a real frequency of 10 kHz is modelled by a frequency of 500 kHz.

If, instead of making the internal surfaces of the model perfectly reflecting at all frequencies, the surface impedance and structural absorptions of the proposed structure are modelled, then additional quantitative measurements such as reverberation time, early-to-late energy ratios and other more or less meaningful objective criteria can be made. References 6 to 12 are given as examples, though they do not constitute a comprehensive bibliography. However, at scale factors greater than about 20:1, the selection of suitable materials for the modelling of surface or structural absorption over a wide frequency range becomes increasingly more difficult and the benefits obtained from further reductions in the size of the model become less, even for models of very large structures. The absorption of the air or other fluid inside the model must also be scaled appropriately (see Section 1.2).

In broadcasting studios, unlike concert halls, the main requirement is that the sound quality received by a microphone should be satisfactory. This therefore permits in addition to the objective measurements, a meaningful subjective assessment. Such subjective assessments are possible for selected seat positions in models of concert halls but are complicated by the requirement to model both the effects on the sound field of the listener's body and the binaural hearing process itself, a matter which has not been completely resolved even at full-scale. In the model of a broadcasting studio, the sound from the model as recorded by a representative microphone arrangement can, after reversing the scaling process, be judged as if it had originated in a full-size studio.

To make these subjective assessments using scaled recognisable real signals such as human voice, solo instruments and full orchestra, requires that three preliminary conditions be satisfied. First, the test signal must not be contaminated by any acoustic qualities of its own. For real signals, this means that it must originate in a space equivalent to a free-field[13]. Second, a means of scaling the frequency spectrum whilst retaining a reasonable sound quality in terms of signal-to-noise ratio and distortion must be available. Third, the means of reproducing and recording the scaled acoustic signals within the model, also with reasonable quality in terms of signal-to-noise ratio, distortion and freedom from colorations, must be available. These electro-acoustic transducers ideally must also reproduce the directionality of the source and of a typical microphone respectively[14]. The development of new improved instrumentation to satisfy these conditions is the subject of a companion report[15].

The difficulties associated with meeting these requirements limit the practical acoustic scaling factors to about 10:1 if a useable frequency range appropriate to music rather than speech is to be achieved. In the earlier work by the BBC on acoustic scale modelling[14,16–21] a useable frequency range of 50 Hz to 12.5 kHz was achieved at a scale factor of 8:1, corresponding to a model frequency range of 400 Hz to 100 kHz in air.

1.2. Air absorption

To maintain the proper acoustic geometry, the sound energy absorption as a function of distance in the medium within the model must be scaled. For normal atmospheric air the increase in attenuation per unit distance is greater than proportional to the frequency in the frequency range of interest in acoustic scale modelling. In normal, moist air two mechanisms for the absorption of sound energy are present. The first is fundamental to the propagation of sound through any fluid and is a function of the fluid viscosity and the thermal conductivity. For any fluid at a given temperature and pressure, this loss sets a minimum limit. In moist air, an additional loss occurs as a result of the interaction of water vapour and oxygen molecules[22,23,24,25]. Under normal ambient conditions and at normal room acoustic frequencies, this additional loss greatly exceeds the fundamental loss. Two different strategies are therefore available for the reduction of the excess loss, to reduce either the oxygen content or the water vapour content. Both are inconvenient and expensive. The oxygen could be removed by filling the model with pure nitrogen (or other suitable oxygen-free gas) but, apart from the expense of the gas, difficulties would be encountered with seals and with the breathing apparatus needed by personnel carrying out work inside the model. In comparison, removing the water vapour is both more convenient and probably cheaper.

For acoustic scale modelling it is not merely necessary to reduce the sound energy loss per unit distance but to obtain the scaled characteristic for the loss as a function of frequency at model frequencies as exists in normal room air at ordinary frequencies[26]. Figure 1 shows a plot of the normal air absorption as a function of frequency for 50% and 60% relative humidity (the normal controlled range for broadcasting studios) and the scaled* absorption parameter as a function of the scaled* frequency for air with relative humidities of 1% and 2% (the lowest practicable limit of an air-drying system for a moderately large acoustic scale model). The similarity between the real and scaled characteristics is remarkable and purely coincidental. It is not so for scaling factors other than near to 8:1 (see Appendix 1).

Thus in all of this work, before any measurement or listening tests (recordings) are made, it is

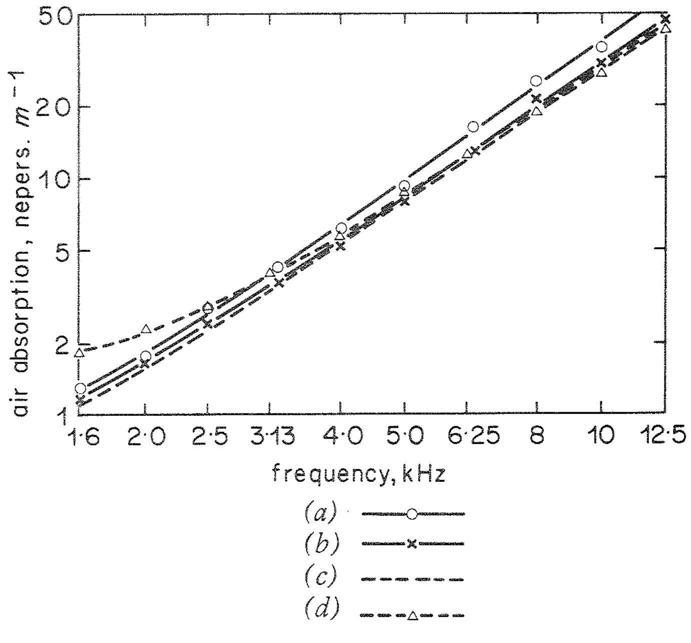

Fig. 1 – Comparison of scale and full-size air absorption characteristics. (a) Full-size, 50% RH, (b) Full-size, 60% RH, (c) Scaled, 1% RH, (d) Scaled, 2% RH.

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*The model absorption parameter and frequency have been scaled to reduce them to their equivalent full-size values (see Section 4.1).

necessary to dry the model and the air within down to a relative humidity of between 1% and 2%.

1.3. Air drying

The relative humidity of the air inside the model can be reduced by circulating the air continuously through some hygroscopic material. The moisture content of the structure and solid contents of the model is reduced by evaporation through the surface finishes. This is a relatively slow process so that the first attempt to dry the model takes a considerable time. However, the reabsorption of the moisture vapour by the structure, for example, from personnel working inside the model carrying out modifications, is also a slow process and subsequent attempts to dry the model take less time, provided that the interval between the two attempts is reasonably short.

The hygroscopic drying agent must have a water vapour pressure at room temperature less than the water vapour pressure in air at 1% relative humidity. For practical reasons, it must be in a convenient form for extracting water vapour from considerable quantities of air and be non-corrosive and non-toxic. For economy, it must also be capable of having its absorbing power regenerated by some convenient and cheap method.

During the earlier work, an air drying system based on the surface absorption by a synthetic zeolite had been developed[14,15]. This drying system consisted of a tank containing about 100 kg of the synthetic zeolite in a granular form through which the air from the model was circulated by a fan and a system of ventilation ducts in a similar arrangement to that used for the ventilation of a full-size rooms. The absorbing power of the zeolite was regenerated by heating and purging the tank with normal room air. This drying system was reasonably effective but, to reduce the rate of ingress of atmospheric moisture, it had been found necessary to surround the acoustic model with a sealed polyethylene tent which was itself maintained at a lower relative humidity than the ambient[14]. Preliminary tests, during the present phase of work showed that the heaters used to regenerate the zeolite were inadequate, the temperature rise being insufficient to drive-off the tightly bound water molecules. An increase of the heater power from 3 kW to 18 kW gave a satisfactory temperature rise (up to 250°C in two hours). The effectiveness and useful life of the regenerated zeolite were greatly improved and the time required for regeneration was reduced from about 16 hours to about 3 hours, excluding the cooling time. With the improved drying system, the polyethylene tent was not required. Using an air-flow rate of about four air-changes per hour, the entire model could be dryed from normal ambient to an extract air relative humidity of 1% in two days of 8 hours each. Subsequent drying periods after installation work inside the model taking one or two days, were only about 1–2 hours. Furthermore, the regenerated zeolite would dry the model between four and eight times before the vapour pressure became so high that the drying time was significantly extended.

1.4. Measurement of relative humidity

The accurate measurement of relative humidities of about 1% in a relatively small, closed volume is difficult. Three possible measurement techniques were considered. The first, using commercially available electronic humidity meters appeared to be simple and accurate but had been rejected during the earlier work because of instability and unreliability[14]. The second technique considered, a dew-point measuring method, was initially rejected as impractical for the dew-points to be encountered ($\simeq -40$°C) but was later reconsidered (see below). The third method, using the traditional wet and dry bulb hygrometer, was the method finally selected in the earlier work. Even this method has serious disadvantages. The main one being that, at relative humidities of 1%–2% at room temperature, the temperature difference between the wet bulb and the dry bulb of about 14°C must be measured to ± 0.1°C in order even to discriminate between 1% and 2% relative humidity. Using two (large) precision mercury-in-glass thermometers such accuracy is difficult to achieve as the individual reading errors are additive. With such large thermometers care must be taken to reduce the effects of radiant heat from the surroundings and especially from the observer taking the readings. The second serious disadvantage is that the entire volume of the model, 16 m^3 in this case, contains only 2.7 g of water at 20°C and 1% relative humidity. The essential evaporation of water from the wet bulb thermometer can add significantly to this total even if most of it is trapped during the first pass of the extract air through the drying tank. The problem is exacerbated by the use of precision mercury thermometers which have large mercury bulbs and are consequently slow to respond and require the extraction of a relatively large quantity of thermal energy (that is, the evaporation of a relatively large quantity of water).

This method was only marginally acceptable in the volume of the model studio. In the model of an ISO standard (200 m^3) reverberation room which was used for the acoustic tests on materials and fittings for the model studio (see Section 1.5), the wet-and-dry bulb hygrometer was totally inadequate for the measurement of low relative humidities.

Even with pre-cooling of the wet bulb, the additional water-vapour introduced into the total volume of less than 1 m³ made it impossible to achieve adequately low measured relative humidities. For the purpose of materials measurements it is not necessary to achieve properly scaled air absorption characteristics as the actual air absorption can be allowed for in the calculations provided that data is available for air absorption coefficients at low humidities. However, the errors introduced can be very large unless the relative humidity is reasonably close to 1% because the air absorption dominates and the differences resulting from the addition of the material being tested become second-order effects.

To reduce these difficulties, a dual electronic thermometer was designed which could be operated as a wet-and-dry bulb thermometer. It used two small temperature-sensitive elements, one of which was attached to a small water reservoir by a conventional wick. An associated electronic circuit mounted at a convenient height remote from the sensor head was based on two ordinary digital panel meters. It gave direct digital readings of the dry-bulb temperature and the dry-to-wet bulb temperature difference with displayed resolutions of 0.1°C and 0.01°C respectively. Calculations based on the data for the temperature sensors and the panel meters gave worst-case errors of ±0.2°C and ±0.04°C respectively over a reasonable range of temperatures for this application. After initial alignment, extensive comparisons with mercury thermometers in normal room air showed no systematic errors. Because of the small size of the temperature sensors, the wet-bulb evaporation requirements were reduced by about 20 times. To avoid introducing excessive water vapour into the model, the hygrometer head was normally removed from the ventilation system. A period of about two minutes was required for the wet-bulb temperature to stabilise after insertion into the extract airflow at 1% relative humidity. This improved hygrometer was satisfactory for use in the model studio but was still barely adequate for use in the model reverberation room.

The method of measurement of relative humidity using a dew-point hygrometer has recently been re-assessed because of these difficulties. One of the significant advantages of dew-point measurement is the sensitivity of the dew-point to changes in the humidity at low humidities. This is in contrast to the other methods which become less sensitive at low humidities. The instrument is based on the cooling of a metal mirror to the point at which dew (in this case, frost) just forms. The mirror temperature is controlled to this point using an optical reflection system to sense the formation of dew. A measurement of the mirror temperature to an accuracy of only ±5°C gives 1% resolution of relative humidity at the humidities used for this work. With thermoelectric Peltier coolers becoming readily available, the controlled cooling of the mirror has become a relatively simple and convenient matter requiring only a suitable power supply.

Unfortunately, no time was available to investigate the dew-point measurement technique further. In any future work on acoustic scale modelling, the development of such a method would be a worthwhile investment.

1.5. Measurement of materials

To construct an acoustic scale model of the interior of any structure, the acoustic performance of the materials used for the model construction must be a scaled version of the full-size materials. In this context, if the dimensions of the materials and the sound energy absorption as a function of frequency are both exact scaled versions of the full-size then the completed model must be a true representation of the way in which sound energy behaves in the full-size structure. The dimensional details need only be accurate down to a size small in comparison with the shortest wavelength being considered. For a model on the scale 8:1 and a maximum frequency of 12.5 kHz, this requires a model dimensional tolerance of $\frac{3}{4}$ mm ($=\frac{1}{4}$ wavelength at 12.5 kHz, scaled). Details smaller than $\frac{3}{4}$ mm may be omitted without impairing the absolute accuracy of the model. For many purposes, such an absolutely accurate model is unnecessary. Part of the purpose of previous modelling exercises has been to establish the degree of accuracy necessary for effective modelling of the acoustics.

The acoustic absorption performance of materials and structures is usually measured in a chamber which itself has little sound absorbing material inside, commonly called a reverberation chamber. The effect of the material under test is assessed by measuring the reverberation time of the chamber both with and without the material under test. During the earlier work on acoustic scale modelling a model on a scale of 8:1 of a standard ISO (200 m³) reverberation room together with its associated air-drying equipment was designed and constructed[27]. Apart from the humidity related problems outlined above the system proved to be adequate.

Reverberation times, both in the model studio and in the model reverberation room, were measured using the equipment developed during the earlier work[14].

2. Earlier work on acoustic scale modelling in the BBC

2.1. Previously reported work

The fundamental principles of acoustic scale modelling as applicable to the BBC were investigated during the period up to about 1974[13,14,16,17,18,19,20]. This work culminated in an 8:1 scale model of the then largest music studio in the BBC, Studio 1 at Maida Vale. The primary reason for deciding to model this particular studio was the general dissatisfaction with the tonal quality of the studio, particularly the 'string-tone' which lacked 'warmth'. The conventional wisdom at the time suggested that, as the studio was considerably lower than was considered desirable for the volume, the ceiling height should be raised by about 4.5 m. In 1969 when this proposal was made, the cost would have been of the order of £250,000 (about £2 million today). The model of the studio as it then existed resulted in a measured reverberation time close to that of the real studio and a sound quality which was judged subjectively to be a reasonable representation of that obtained in the real studio. In fact, the expert listeners concluded initially and quite correctly that the model studio had no orchestra present. After the addition of a model orchestra, the disliked hard 'string-tone' of the studio appeared in the model. This gave sufficient confidence in the modelling technique to justify the investigation of the effects of raising the studio ceiling. The ceiling was raised in two stages by the equivalent of 4.9 m (16 feet) and 2.4 m (8 feet), increases of 60% and 30% respectively in the total volume of the studio. The subjective effect of the additional volume on the reverberation room was pronounced but, after sufficient acoustic absorption had been introduced to restore the reverberation time to the original characteristic there was no discernible subjective difference between recordings made at the three different ceiling heights[18]. Thus, the expensive mistake of raising the studio ceiling was avoided. Further experiments in the model identified the probable cause of the problem and changes to the acoustic treatment and orchestral layout which reduced its significance[19].

Following this work with the model of Maida Vale Studio 1 and taking the generally satisfactory results as confirmation of the validity of the technique, a model was constructed of a proposed new music studio for Manchester (subsequently to become Manchester, Studio 7)[20]. The general principles of the preliminary design were shown to be satisfactory. Following from the experience with Maida Vale Studio 1, a number of different ceiling heights were experimented with. It was again found that, at least for the sound quality as recorded by a microphone, the ceiling height was not a significant parameter. A number of different average reverberation times were also assessed, ranging from 1.9 to 2.75 seconds. The surprising result obtained was a distinct preference for longer reverberation times than had been thought desirable for a studio of this volume. Even the longest reverberation time tested (2.75 seconds) was judged more favourably than the previously assumed optimum of 1.9s. The final choice of 2.25 seconds, based on both subjective preference and on operational requirements, was made and the studio design altered to give this figure. A second important discovery made during this work was that the modular acoustic treatment and especially the low-frequency modules, was much more effective in the large space of the studio (approximately 8100 m^3) than in the reverberation room (200 m^3)[20].

These two factors together reduced the quantity of acoustic treatment which was going to be required in the studio and, by allowing more accurate ordering of materials saved a considerable amount of money. The financial savings made in this aspect alone offset a large proportion of the costs of the entire modelling work for the studio.

2.2. Outcome of the Manchester Studio 7 modelling

Since the completion of the final report on the acoustic scale modelling of Manchester Studio 7[21], the studio has been completed. The main features of the model were all retained in the design. The shape and size of the studio, the disposition of the diffusing ribs, the type and distribution of the acoustic treatment and the design reverberation time (2.25s) were all as had been modelled.

As a further development of the work on acoustic scale modelling and to take advantage of a rare opportunity to observe the progress (acoustically) of the construction of a large music studio, many visits were made to the construction site to measure reverberation times inside the studio as the acoustic treatment was installed*.

The first of these visits, when the shell was completed but without any intentional acoustic absorption showed that the low-frequency reverberation time was much shorter than had been expected. The second visit to the site took place after the initial installation of the low-frequency modular absorbers and the third after the addition of the first phase of wide-band modular absorbers*. At this stage it was evident that there was a great deal of excess low-frequency absorption, giving a reverberation time of little over 1 second at 50 Hz, and insufficient high frequency absorption, giving a reverberation time of 3 seconds at 2 kHz.

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*The detailed results from these visits will be included in later sections of this report to avoid repetition and confusion of the general discussion.

Calculations based on the measured performance of the modular absorbers in the studio showed that it was necessary to remove nearly all of the low-frequency absorbers, sufficient absorption being provided by the so-far unproven cause of the excess absorption of the 'empty' shell and the residual low-frequency absorption of the high-frequency absorbers. Instead**, the front panels and contents of the low-frequency absorbers were removed and replaced by a front panel of 25 mm blockboard. This front panel was covered with a suitable floor carpet to increase the high-frequency absorption to assist in the reduction of the high-frequency reverberation time. A further measurement showed that the reverberation time characteristic was much improved but not entirely corrected.

During all of the modelling study, it had been understood that the occasional presence of an audience should be tolerated. Rudimentary model experiments had been carried out to confirm that this would be so. However, in the period between the completion of the model study and the completion of the studio, operational requirements had changed to the point where the presence of the audience would be more than "occasional". A measurement of the reverberation time with the audience seating in the studio showed that the audience seating provided sufficient high frequency absorption to compensate to a large degree for the shortages of such absorption from the fixed acoustic treatment. The studio was subsequently handed over to the users in this condition, with the average reverberation time being lower (1.9–2.05 s) than the design figure, with a slight rise in reverberation time between 1 kHz and 2.5 kHz and with a large proportion of the high frequency absorption being concentrated in a relatively small patch on the floor, i.e. the audience seating.

The initial reactions of the users were favourable and the sound quality as recorded by normal microphone techniques was both very good and as had been predicted from the model. However, from the beginning of its use by the musicians, some reservations were expressed about the sound quality for listeners within the studio. These persisted during the familiarisation period and so further changes were made to the acoustic treatment to reduce the high frequency reverberation time and to introduce some asymmetry. In particular, the wall facing the orchestra was made much more absorptive in order to reduce the amplitude of the first reflections from this wall. Smaller changes were made to the walls near to the orchestra to increase the amplitude of the early reflections and thus assist the musicians in their timing. To achieve these modifications the users had to forego the ability to change the orientation of the orchestral layout, a desired option that had been important at the design stage.

3. Reasons for re-evaluation of the Manchester Studio 7 model

Although the completed studio reproduced many of the features of the model, there were inevitably some structural details which were changed in the period between the initial proposal on which the model had been based and the final, detailed construction drawings. Most of these changes had no acoustic consequences but there were some details of the interior of the studio which had not been included in the model study. When the studio shell was completed it was immediately apparent that there was an excess of low-frequency structural absorption. This excess had consequences on the whole of the remaining acoustic design and was responsible, ultimately, for the average reverberation time being shorter than the design figure. The cause of the excess absorption was in dispute but the wooden-faced ventilation trunking was the prime suspect. One reason for reconstructing and re-evaluating the model was to determine the cause of the excess low-frequency absorption. To this end, a somewhat different philosophy for the model design was adopted. Instead of attempting to model only the absorption of the interior surfaces as for the first model[20], the new model was to be constructed as a scaled structure with all of the relevant materials being as near as possible to structurally and acoustically scaled versions of the materials actually used in the studio rather than simulations of the full scale absorption characteristics. The reverberation time results obtained in the real studio during the construction and the installation of the acoustic treatment were to be used to check the behaviour of the components of the model during its construction.

The second main reason for reconstructing the model, the original model having been destroyed in the intervening period, was the prospect, at the time, of a major new BBC concert hall in the new London Broadcasting House. It seemed likely that the acoustic scale modelling technique might contribute to the design. In the period since the original modelling work was completed (1974), some advances in the technologies of loudspeakers, microphones and amplifiers gave reason to believe that the recorded sound quality from an 8:1 scale model might be significantly improved. The new model was intended to permit testing and evaluation of these advances preparatory to using them in other models. This electronic and electro-mechanical development is the subject of the companion report[15].

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*For both acoustic and practical reasons, the treatment was installed in several stages. The initial installations comprised 75% of the anticipated requirement.

**The Acoustic Consultant recommended this alternative course of action based mainly on aesthetical and practical factors.

The third reason for the reconstruction of the model, the study of the correction of some defects in the real studio, was overtaken by events at an early stage. The corrections were applied by normal acoustic design procedures without the aid of an acoustic scale model. Indeed, most of these defects were related to the perceived sound quality within the studio, by performers and audience, features which are not at present amenable to study using an acoustic scale model (Section 1.1).

4. Construction of the model and the results obtained

4.1. General

In this section, a number of results are presented for reverberation time and absorption coefficients for both the model and the real studio. To simplify the presentation and to permit direct comparison with the results from the full-scale, all of the model results have been converted to a full-scale frequency range. Also implicit in these results is that measurements of model reverberation times are carried out at a relative humidity of 1%–2% to obtain the (nearly) correct scaling factor for the air absorption.

4.2. Shell

The studio dimensions are 26.6 m × 22.2 m × 14.025 m high, giving a gross volume of 8282 m^3. The walls and ceiling of the real studio are constructed from cast, reinforced concrete with a plastered and painted inner surface to a total thickness of 300 mm. The diffusing ribs which form a feature of this studio are pre-cast concrete sections bolted through the walls. Such a structure should have a very low absorption coefficient at all frequencies. The floor of the studio consists of 25 mm thick, varnished wood strips on battens set into the top screed of the concrete floor. In modelling the empty shell, the aim was to obtain as low an absorption coefficient as possible at all frequencies to provide the enclosure within which the remainder of the work could be carried out. Also, although the sound insulation of the model structure was not part of the investigation, it was necessary to achieve reasonable sound insulation to exclude the effects of the room in which the model was constructed.

Measurements of absorption coefficients using the model reverberation room with the top panel replaced by the test panel showed that dense, 25 mm thick chipboard (particle board) with a hard paint system* gave lower absorption coefficients than any other reasonably cheap material. The model shell was constructed in sections following the natural lines of the interior reinforcing piers to simplify construction and to allow for subsequent storage. The 14 wall sections and three roof sections were bolted together on a base of 25 mm chipboard. Figure 2 shows an intermediate stage in the construction.

The ventilation of the model had to have reasonably uniform air distribution because of the drying requirements. It was therefore designed to be similar to the real studio but with fewer inlets and outlets for simplicity. In the real studio the ventilation plant is at ceiling level with ceiling-level supply and floor-level extract. The vertical extract ducts were within the studio. In the model, these vertical extract ducts were not functional for two reasons. The first was that, as they were believed to be the origin of the significant excess low frequency absorption in the studio, it was desirable that they be easily replaceable. This was made easier by not having to consider their ventilation function. Second, it was more convenient in the model ventilation system to have a floor-level peripheral extract

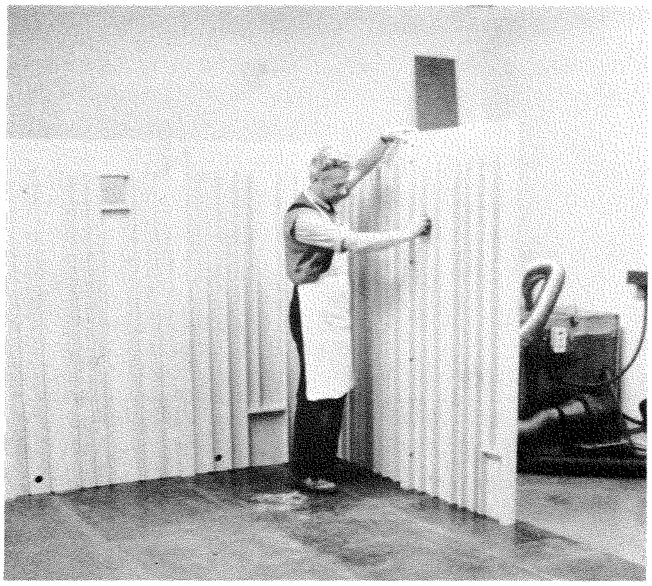

Fig. 2 – Construction of the scale model.

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*Trimite – Primer – 1 pack epoxy base, AP15; U/coat – QU50; Top coat – Q50.

duct. Figure 3 shows the general appearance of the exterior of the model with the ventilation system installed.

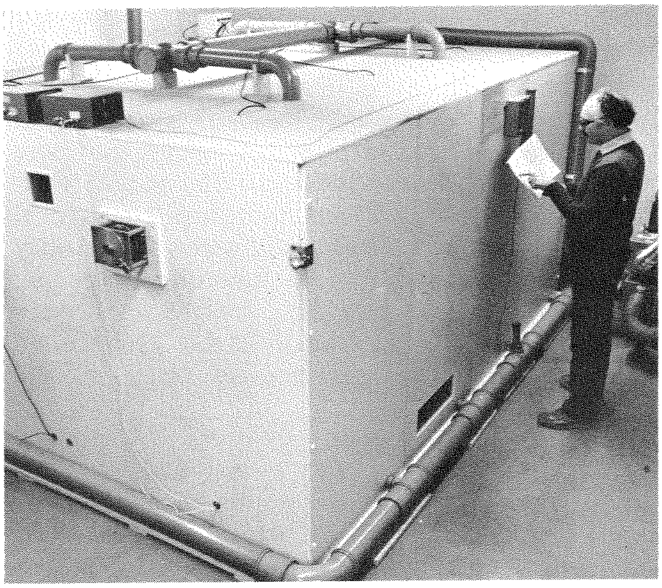

Fig. 3 – Exterior of model showing arrangement of ventilation trunking.

The reverberation time of the empty model shell was as shown in Fig. 4(a). Figure 4(b) shows, for comparison the calculated reverberation time of the empty studio with contributions to sound absorption by the painted and plastered concrete walls and ceiling, the wood-strip floor and the air. The studio was not measured in this condition; the ventilation trunking was already installed at the time of the first tests in the studio.

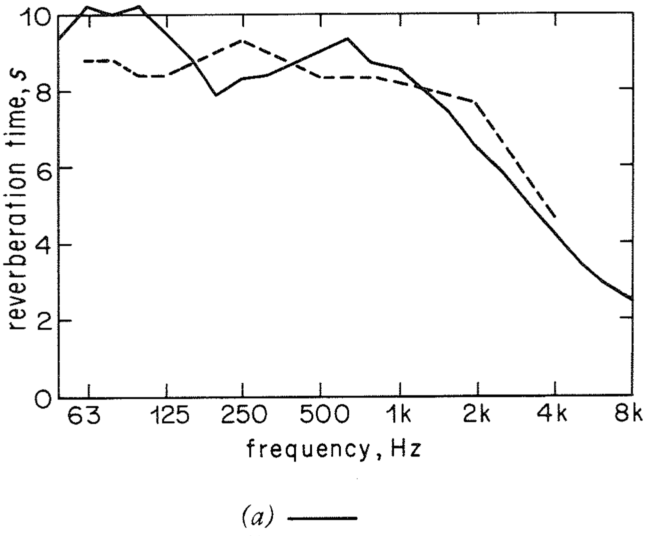

Fig. 4 – Reverberation time of empty studio. (a) Model, measured (b) Studio, calculated.

4.3. Ventilation trunking

In the studio, the extract ventilation trunking covers consist essentially of 25 mm thick blockboard over an airspace 785 mm wide and 175 mm deep. They run the full height of the studio, that is 14 m, and number 28. Dimensional analysis of the resulting resonant system formed by the mass of the front panel and the combined compliance of the enclosed airspace and the inherent panel stiffness (Appendix 2) shows that the resonant frequency will be scaled by the scaling factor if all dimensions are reduced by the scaling factor and the density and Young's Modulus of the front panel are unchanged. Material properties for timber are highly anisotropic, virtually unobtainable and in any case vary within wide limits. Based on some data originally collected for the purpose of loudspeaker cabinet design[28] it was thought that 3 mm, 3-ply plywood would be a reasonable model of the 25 mm blockboard if the panels were cut with the outer layer grains running parallel to the long axis (i.e. the height) of the trunking. The Young's Modulus is important in this application because the inherent stiffness of the panel is the dominant component of the total compliance (Appendix 2).

Two different designs of model ventilation trunking were tested. The first was constructed entirely of 3 mm plywood as completed box sections. These were stuck into the appropriate places on the model walls with self-adhesive pads. The reverberation time of the model with this ventilation trunking is shown in Fig. 5(b) together with that for the studio in a comparable condition, Fig. 5(a). Although the low-frequency reverberation time of the empty shell (Fig. 5(d)) was reduced significantly and was close to that of the real studio up to a frequency of 315 Hz, the ventilation trunking also introduced a large amount of unwanted high-frequency sound absorption. This occurred despite the trunking being carefully and heavily varnished to reduce the h.f. absorption.

A second model ventilation trunking was constructed which more accurately modelled the studio installation. This comprised the front panel only fixed to softwood fillets which were themselves fixed to the fins. This modification was done mainly to eliminate the narrow gaps between the sides of the original boxes and the fins, gaps which were thought might be responsible for the unwanted high frequency absorption. Figure 5(c) shows the reverberation time of the model with this modified ventilation trunking. Again, the low-frequency reverberation time is close to that of the studio although not as close as the first ventilation trunking. However, the absorptión of this second attempt at frequencies

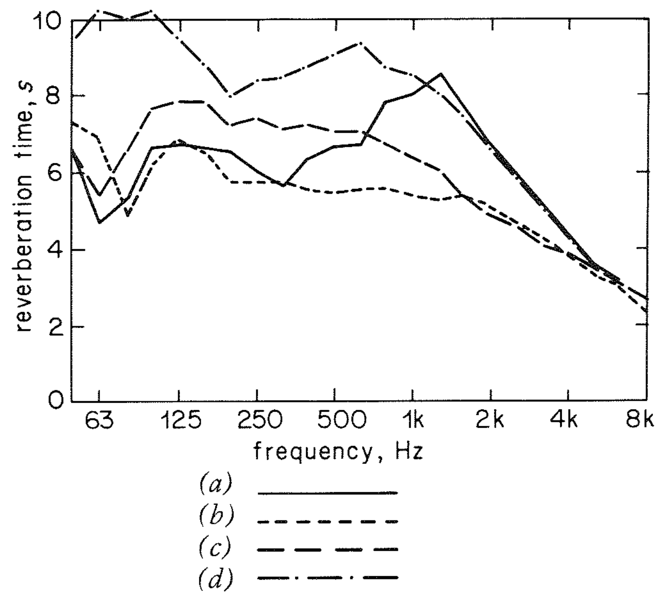

Fig. 5 – Reverberation time of studio with ventilation trunking. (a) Studio, corrected to 60% RH, (b) Model with first ventilation trunking, (c) Model with second ventilation trunking, (d) Model without ventilation trunking.

above 1.6 kHz was essentially the same as the first attempt. In neither case is the high-frequency reverberation time of the studio in a comparable condition modelled accurately.

The cause of this unwanted high frequency absorption was not discovered by extensive testing of the component parts in the model reverberation room and it was decided that as it formed a reasonably small fraction of the total absorption which would ultimately be installed in the model the work should continue on the remaining aspects.

It is possible that the studio ventilation trunking behaved in the same way at high frequencies and that the origin of the difference is not in the ventilation trunking but in the residual absorption of the empty shell. For this to be so the studio without the ventilation trunking would need to have a much longer reverberation time and the calculation of this reverberation time would need to be incorrect. In this case, the performance of the model ventilation trunking would be correct but the reverberation time of the empty model shell would be too short. Without a measurement of the bare studio shell reverberation time there is no way of separating these two possibilities.

The effective absorption coefficients of the model ventilation trunking may be calculated from the results of Fig. 5(b), (c) and (d) and those of the real ventilation trunking estimated from the results of Fig. 4(b) and Fig. 5(a). Figure 6 shows the resulting absorption coefficient characteristics for the studio ventilation trunking, Fig. 6(a), and the two model ventilation trunkings, Fig. 6(b) and (c).

At low frequencies, both of the model ventilation trunkings have reproduced the peak in absorption at 63–80 Hz. The first attempt has a resonant frequency which is $\frac{1}{3}$ octave too high in frequency. However, the second attempt has, over the range of frequencies 50 Hz to 125 Hz, modelled the studio ventilation trunking accurately with the same resonant frequency and a similar peak absorption. Over the frequency range 125 Hz to 315 Hz, the first attempt is a more accurate model of the studio. Above 315 Hz neither attempt is a particularly good model.

4.4. Low-frequency modular absorbers

As described in section 2.2, after the initial installation of the acoustic treatment in the studio, the low frequency modular absorbers were modified to reduce their low frequency and increase their high frequency absorption coefficients. The 25 mm thick blockboard which was used to replace the original perforated hardboard front covers was intended virtually to eliminate the low frequency absorption. Unfortunately, the blockboard panels have a fundamental resonance at a similar frequency to that at which the original acoustic treatment had had a maximum absorption coefficient. The increase in the front panel mass had been offset by a corresponding decrease in the system compliance, that is, the combination of the panel and enclosed airspace compliances. Therefore they behaved as somewhat inefficient panel or membrane absorbers and the reduction in low-frequency absorption coefficient was not as great as had been hoped.

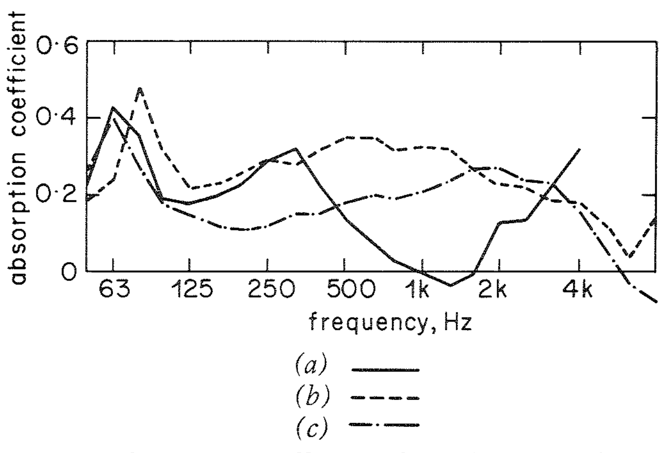

Fig. 6 – Absorption coefficient of ventilation trunking. (a) Studio, from calculated empty reverberation time, (b) Model, first ventilation trunking, (c) Model, second ventilation trunking.

The structure of the final versions of the low frequency modular absorbers was modelled using the same 3 mm, 3-ply plywood to represent the 25 mm blockboard as was used to model the ventilation trunking. The side and rear panels of the modular absorber boxes in the studio were made of 9 mm plywood and 6 mm hardboard respectively. It has been shown[29] that these panels can have significant sound absorption but it was not practicable to use the 1.1 mm plywood and 0.75 mm hardboard required properly to model these components. Instead 1.5 mm thick acrylic sheet (Perspex) was used for ease of construction*. Appendix 3 indicates that the side panel resonance of these modules may be nearly one octave lower than it should be.

Initially, the model low frequency acoustic treatment was installed without the additional high frequency absorption provided in the studio by the carpet covering on the front surface to reduce the difficulty of interpreting the results. The first installation, with the 3 mm plywood front panels securely glued to the side panels, showed very low absorption coefficients at all frequencies. In comparison with the full size modules in the studio, the front panels were fixed too securely. A second attempt with the front panels fixed by a flexible joint more like the comparatively loosely fixed front panels of the full-scale modules resulted in measured absorption coefficients in the model studio as shown in Fig. 7(b). The comparable measurement was not made in the studio because low frequency treatment was changed to its final form late in the construction schedule and after other acoustic treatment had been installed. Therefore, the absorption coefficients of the final low frequency acoustic treatment in the studio can only be derived by an indirect route from the final measurement of reverberation time in the completed studio. Figure 7(a) shows the results of this process. Because of the number of steps involved, this estimate is not very reliable. It was, however, the guide used during the development of the modified low frequency model modular absorbers.

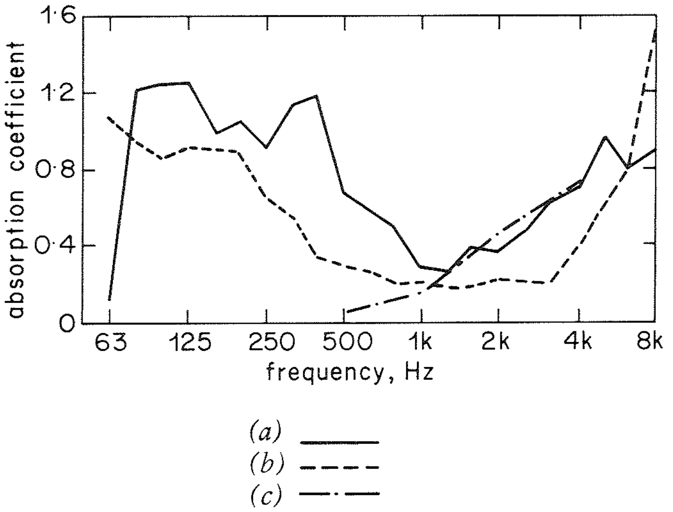

Fig. 7 – Absorption coefficient of low-frequency modular absorbers. (a) Studio, derived indirectly from other measurements, (b) Model, measured, (c) Manufacturer's figures for carpet covering.

Considering the possible errors in the derivation of the studio modules' performance, Fig. 7 shows a moderate match between model and full size modules. The general shapes of the characteristics are similar but the models seem to have overall a lower absorption coefficient. It is possible to interpret in Fig. 7 the effects of the side panel resonances. As already mentioned, it can be shown that the model construction would result in approximately a one octave error in these resonances. Measurements on the low frequency modular absorbers in the studio showed a primary side-panel resonance at 250 Hz. In the model this resonance should therefore occur at about 125 Hz. It is just possible to see in the results of Fig. 7 differences in absorption coefficients corresponding to these two frequencies.

The reason for the increase in absorption coefficient of the studio modules at frequencies above 1 kHz is the front covering of carpet. The shape and frequency range of the increase corresponds very closely with the manufacturer's figures for the absorption coefficient of the carpet Fig. 7(c). The reason for the sharp rise in absorption coefficient of the model absorbers above 3.15 kHz is not known. Such a large increase in absorption coefficient must have a substantial origin. Moreover, such an increase was not evident in any of the preceeding tests on the low frequency modules and their component parts in the model reverberation room. Neither is it an artifact of the relative humidity. In acoustic scale model work, that is at relative humidities of around 1–2%, the humidity effects and corrections are most significant at real frequencies of about 5–20 kHz, equivalent in the 1/8 scale model to frequencies of 0.6 to 2.5 kHz at the full size. Unlike ordinary acoustic work, the largest humidity-related errors do not occur at the highest frequencies. This matter remains unresolved.

4.5. Wide-band modular absorbers

The same model wide-band modular absorbers were used as in the previous work[20]. These consisted of 1.5 mm acrylic sheet boxes with 3 mm thick dense felt to model the 25 mm thick mineral wool and a

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*The same model modular absorber boxes were used as in the earlier work for the wide-band modules.

20% perforated* brass front panel. In the case of these absorbers, the front panel material was not thought to be acoustically significant. The layer of 0.02 mm thick Melinex film used to cover all of the wide-band modular absorbers in the earlier work and which was initially reproduced in the real studio by two layers of 0.08 mm Melinex film was removed from 75% of the modules to model the final state of the studio. Figure 8(a) shows the average absorption coefficient characteristic of the wide-band modules in the studio and Fig. 8(b) that of the scaled modules in the model.

As in the case of the low frequency modular absorbers (Section 4.3), the performance of the final version of the wide-band absorbers was not measured directly in the studio. An estimate of their performance was extracted in several steps from other measurements. However, Fig. 8 shows that, except for the low frequency region, the estimated performance of these modules is reproduced fairly closely by the models.

4.6. Orchestral rostrum

The design of the studio orchestral rostrum was based on that installed in Maida Vale Studio 1, developed and tested during the earlier model work on the Maida Vale studio[19]. With only minor detail changes, the same rostrum was intended for the new studio. Therefore, for the modelling of Manchester Studio 7, the identical model rostrum was used initially and also in this later work. The acoustic absorption of the rostrum is not large and, in the completed studio only measurably affects the low frequency bands. Figure 9(a) shows the absorption units of the studio rostrum and Fig. 9(b) those of the model rostrum. In both cases, these results were obtained from a pair of measurements in an otherwise nearly empty room, that is, before any intentional acoustic treatment has been installed. Even so, both the studio and the model had the absorption of the ventilation trunking which reduced the "empty" reverberation times at low frequencies, thus also reducing the accuracy of the absorption measurement at low frequencies (see section 5.1).

Over most of the frequency range, the model orchestral rostrum is a reasonable representation of the full size rostrum.

4.7. Audience seating

The audience seating arrangements in the studio consisted of open steel-frame staging in the form of seven wide and shallow steps with a total area of 15.75 m × 6.53 m. The total rise from floor to rear step was 1.23 m. The upper surface was covered with 20 mm flooring grade board which was itself covered with carpet. Approximately 250 free-standing, metal-frame chairs with thick cushions were placed on the structure.

In terms of sound energy absorption, both the open steel structure and the chair frames are likely to be negligible. There will probably be some low frequency absorption from the floorboards behaving as panel absorbers and certainly high frequency absorption from the carpet and the upholstery of the seats. The fabric of the seat covers appeared on inspection to be a close-weave synthetic material which might be somewhat opaque at high frequencies. Also, when the time had come to introduce

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*Approximately, 0.51 mm diameter holes at 1.06 mm square spacing in 0.4 mm thick material.

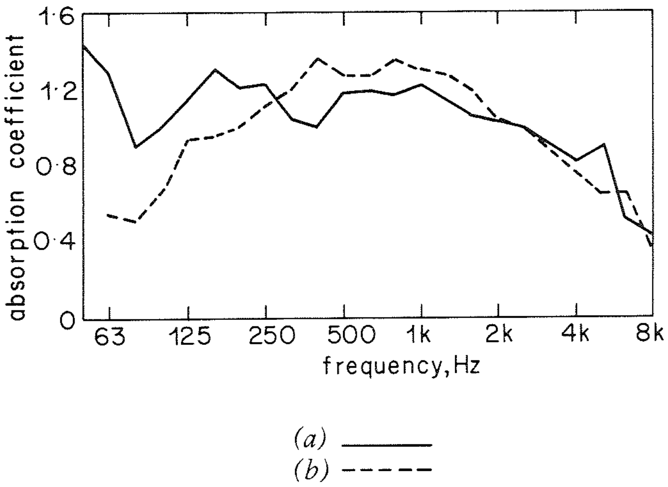


Fig. 8 – Absorption coefficient of wide band modular absorbers (25% of total covered with Melinex film). (a) Studio, (b) Model.

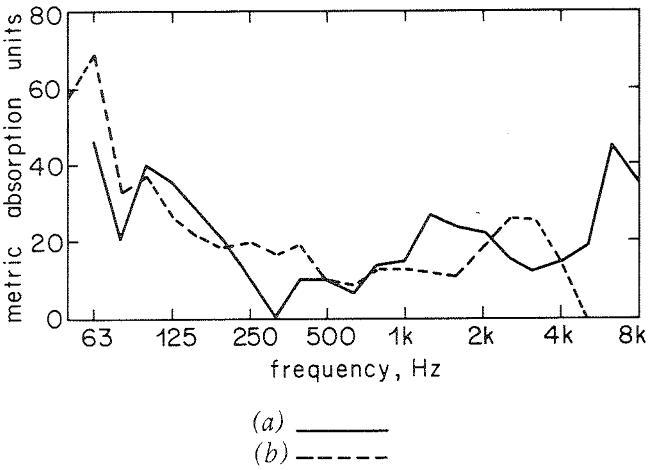


Fig. 9 – Absorption units of rostrum as measured in empty studio. (a) Studio, (b) Model.

the audience seating into the model, the results of a measurement of the effective absorption coefficient of the full-scale audience seating were available. For reasons which will be discussed later (Section 5.1) these results were not very reliable but they did serve as a guide to the design of the model audience seating.

The model consisted of a 3 mm thick plywood sheet structure to represent the boards of the full size staging. This was made to be self supporting with only a minimum amount of framework beneath. The carpet and upholstery absorption was simulated by covering all of the horizontal surfaces with 3 mm dense felt. A further covering of 0.05 mm polyethylene sheet was used to limit the high frequency absorption.

Figure 10(a) shows the absorption coefficient of the full-size seating in the studio and Fig. 10(b) the equivalent result in the model. The studio results are not given below 250 Hz because they were totally unreliable.* Above 250 Hz the model results reproduce the full size results reasonably well although the apparent differences between 400 Hz and 800 Hz are fairly large.

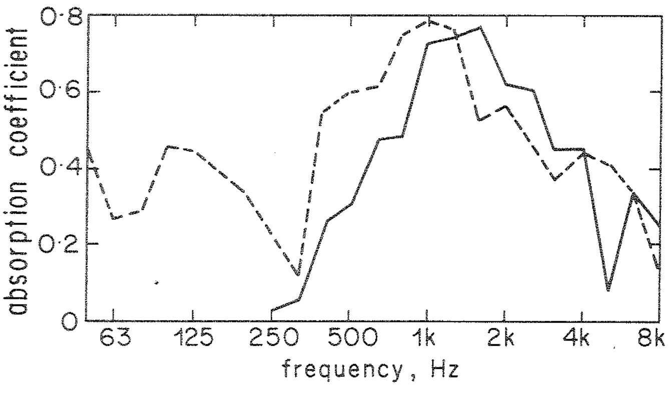
4.8. Orchestra

After the results were obtained showing the effects of the presence of an orchestra in Maida Vale, Studio 1[17], the model orchestra has been included in all subsequent acoustic scale modelling work. The effect on the average reverberation time of the acoustic absorption of even 60–70 orchestral players is small in rooms of such volumes but other, more subtle possibilities[19] require the inclusion of at least the equivalent acoustic absorption properties. Figure 11(a) shows the measured absorption coefficient of a real orchestra in Manchester Studio 7 and Fig. 11(b) that of the model orchestra in the acoustic scale model which is the subject of this report. In neither case were the effects of the musical instruments included. Results for frequencies below 250 Hz are not given because they were both very unreliable and are probably very near to zero absorption anyway. Above 250 Hz, the results are very similar, though subject to marked experimental error.

4.9 Overall results

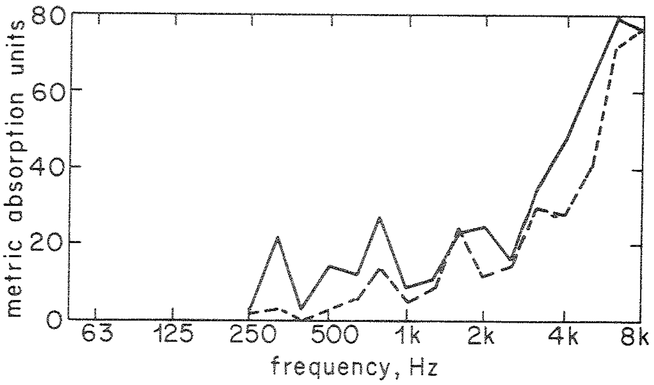
Figure 12(a) shows the reverberation time characteristic as measured in the completed studio. Figure 12(b) shows the equivalent characteristic in the model. In both cases, the studio and model were complete with the final version of the acoustic treatment installed and with the orchestral rostrum and the audience seating present. Figures 13 and 14 show typical views of the interiors of the studio and the model. No orchestral players or audience were included at this stage for the measurements, though they are shown in the photograph of the model interior, Fig. 14.

Though reasonably close for acoustic design work, the completed model reverberation times were too long by about 15% at low and mid-frequencies and increasing to about 20% at high frequencies. However, the general shape of the studio reverberation time characteristics has been reproduced fairly



(a) ———
(b) - - - -

Fig. 10 – Absorption coefficient of audience seating. (a) Studio, (b) Model.



(a) ———
(b) - - - -

Fig. 11 – Absorption units of orchestra. (a) Measured in studio, (b) Measured in model (Both measured in otherwise fully treated studio.)

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*The only direct measurement of the effects of the audience seating in the studio was made using the "pistol shot" method (see Section 5.1). The signal-to-noise ratio of the resulting recordings was insufficient at low frequencies to permit meaningful interpretation. The effects of the audience seating were also derived indirectly in several steps from other measurements. The accumulation of all of the errors introduced by this indirect method (see Section 5.1) resulted in very large overall errors.

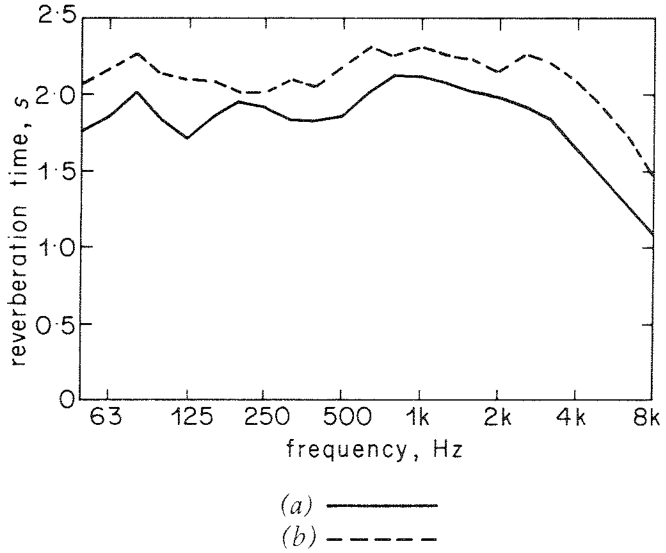

Fig. 12 – Reverberation time of the completed studio. (a) Studio, (b) Model.

well even in the detail of the slight rise in average reverberation time between 500 Hz and about 3 kHz.

The average absorption coefficients were calculated for the total absorption in both the studio and the model. Based on an area of 100 m², which was approximately the total area of all of the different types of absorption excluding the shell itself, these averages came to 0.63 for the studio and 0.54 for the model, a difference of about 15%.

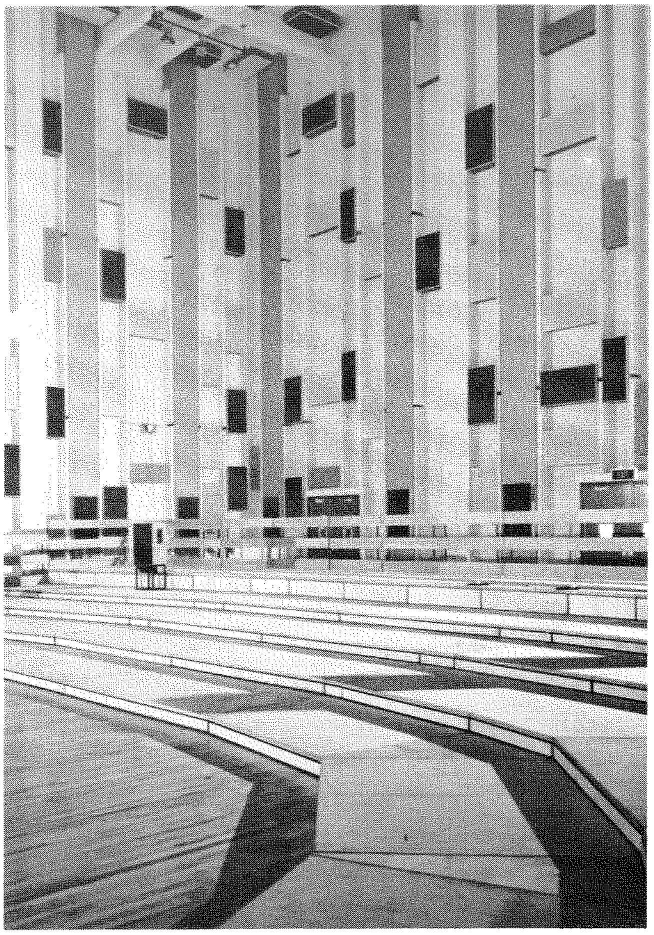

Fig. 13 – Interior of Manchester Studio 7.

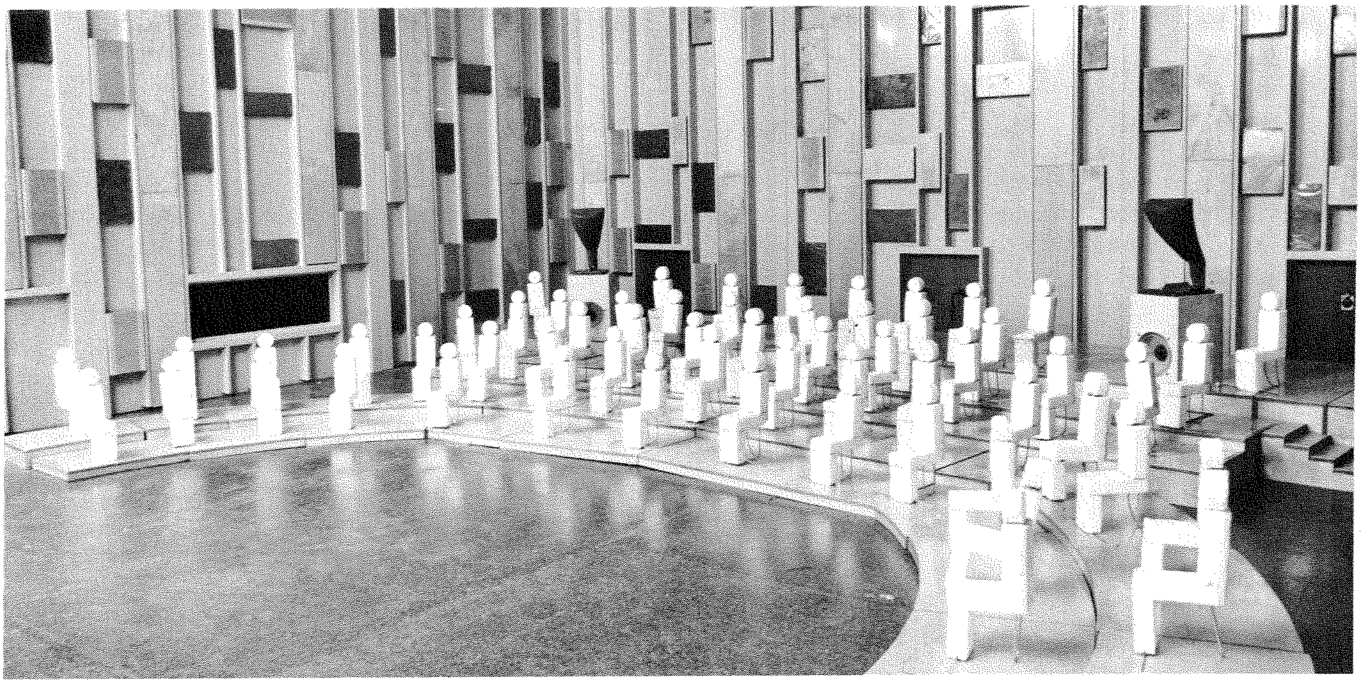

Fig. 14 – Interior of acoustic scale model.

5. Discussion of results

5.1. Errors in measurement

Reference has been made in many places throughout this report to the poor reliability of some of the results. The measurement of reverberation-time, especially at low frequencies, is itself somewhat inaccurate. Even in such a large volume as this orchestral music studio, individual variations between decays are large and meaningful results can only be obtained by averaging a very large number of decays at points distributed throughout the volume of the space. Until the recent development of automated equipment this was a laborious and time-consuming exercise. In such a large space it is also difficult to explore the whole volume adequately, especially vertically because of the height of the space. During the course of the measurements in the studio, two items became available which simplified the measurements. The first was the development of microprocessor-controlled, reverberation-time measuring equipment (ART)[30] which, because it had true screen storage of the decays (rather than a long-persistence phosphor), was much easier to use for relatively long reverberation times. It also had a crystal-oscillator-based time-base, thus eliminating possible calibration errors. Being automatic, it also simplified the analysis of large numbers of decays. The second item was a rotating microphone boom which, by continuous rotation, carried out a spatial averaging process (over a limited space) automatically. Such techniques, if applied carefully, may give results to within ±1–2%. It is unlikely that any of the results from the studio are so accurate. They are however, probably* within ±5%.

The equipment was used for some of the later measurements in the studio. Even when available it could not always be used. For example, when measuring the effects of the orchestra, it was not possible to ask the subjects to remain quiet in a very noisy environment for about one hour whilst the measurements were made. Such occasions were dealt with by the much quicker but very much less accurate method of recording the effects of explosions (in this case, pistol shots) from a number of different locations. The recordings were subsequently analysed in the conventional way. The errors involved in such measurements could be ±10%.

In the model, the automatic equipment could not be used because it did not cover the whole of the required frequency range. A small rotating microphone boom was constructed but it could not rotate continuously and was used merely to move the microphone to the next position between measurements. Even so, the generally more comfortable conditions of a familiar laboratory and the shorter reverberation times of the model contributed to improved accuracy with the older, manually operated equipment. With the exception of one or two calibration errors, the measurement errors of reverberation-time in the model are probably less than 3%.

The errors in the measurement of reverberation times are not of themselves very important or serious. However, to obtain a figure for sound absorption requires the subtraction of two values for the total absorption, each obtained from a measurement of reverberation time. If the total absorption in each case is large and the difference caused by the material under test is small then the errors in the measurement of reverberation times are greatly magnified. For this reason, laboratory measurements of absorption are made in a room which contains little sound absorption except that of the material under test.

The errors in measured absorption coefficient caused in this way can be illustrated by some calculated examples. For the first case, take the example of the model ventilation trunking from section 4.1. This was installed in an acoustically otherwise nearly empty room. At low frequencies the trunking had a relatively high coefficient of sound absorption, 0.4 at 63 Hz. Errors of ±5% in the two reverberation times from which this result was derived lead to a range of answers for the absorption coefficient of 0.458 to 0.349, a total range of about ±13%. In contrast, take the example of the audience seating in the studio. This was installed in an otherwise completely acoustically treated room. At 1 kHz it appears to have an absorption coefficient of 0.73. Applying the same 5% error bands to the reverberation times gives results in the range 0.98 to 0.47, an error band of ±35%. The errors are even worse if the material being assessed has a low absorption coefficient. Taking the same example of the audience seating but at 250 Hz, the range of values of absorption coefficient for a ±5% error in reverberation times is 0.34 to −0.26, with a median value of 0.03!

Thus, in the measurement of acoustic absorption coefficient, errors which can sometimes be very large must be expected and accepted as inevitable.

5.2. Modelling of studio details

Many of the results given for the performance of individual items in the studio show significant differences between the full size and the model, even considering the possible measurement errors discussed in section 5.1. However, in all cases, the general shape of the results obtained show similar trends. Even by normal acoustic standards where exact repeatability is rarely obtained, the modelling appears to be somewhat inaccurate. However, the basis on which this model was constructed was rather different from the usual modelling of the absorption coefficient characteristics. With the sole exception of the audience seating, this model was constructed entirely from an understanding of the physical bases on which the various elements were supposed to work and not from a knowledge of the actual performance of the elements. Given this consideration, the results obtained for the model elements are reasonably close to those obtained in the full size studio.

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*A formal error assessment has not been carried out on all of the results. The figures quoted are estimates and about 90% confidence level.

The sole exception to this approach was the audience seating arrangement. Although a study of the construction was used as the basis of the model design, measured results for the full size seating were used to confirm the model performance. This was done because the seating, and especially the upholstery, represented acoustic absorption of a type not previously encountered in modelling and of which there was no prior experience.

5.3. Conclusions to be drawn from this model

Within the overall limits of accuracy already discussed, this re-evaluation of the modelling technique has confirmed that the ventilation trunking in the studio was the prime cause of the initial excess low-frequency sound absorption. It was also confirmed that the present understanding of the way in which some types of materials behave acoustically is essentially correct. In particular, the low frequency absorption of thin* resonant panels has been verified. In the two separate cases of the ventilation trunking and the front panels of the 'disabled' low-frequency modular absorbers, relatively thick** wooden panels were shown to have high absorption coefficients at low-frequencies.

This re-evaluation has also shown that mechanical and structural considerations can be used to predict the acoustic performance of common types of acoustic absorption at different scales provided that the details of the construction are sufficiently well known and accurately reproduced. One exception to this was the spurious high frequency absorption of the model ventilation trunking covers. Further work is required to confirm this effect and to discover the cause.

The model has also demonstrated that the measurement of the acoustic properties of materials is liable to errors, especially when the room in which the measurement is being carried out contains other acoustic absorption.

5.4. The future of acoustic scale modelling

Most of the arguments and discussions in this report relate to the construction of an acoustic scale model of an existing studio. That, indeed, was the purpose of this work. However, all of the considerations apply equally well to working in the reverse direction, that is, constructing a full-size version of an existing model. It is in this direction that acoustic scale modelling is most commonly thought of and in which the financial justifications for the technique are usually to be found. Throughout the remainder of this sub-section, the terminology of modelling can be considered to be equally applicable in either of these two directions.

The construction principles used for this model are not necessarily the only ones or even the optimum ones. It may be more accurate to model the actual absorption coefficient characteristics of the materials to be used. By rigorously accurate modelling of such a kind, a scaled reproduction of an arbitrarily high degree of accuracy can be obtained. However, it is not always possible to establish the true acoustic behaviour of materials, perhaps because they do not behave in the appropriate way until assembled at the correct size or in the correct sized space. It is also probably uneconomic and to some extent self-defeating slavishly to copy every small detail of acoustic absorption coefficient. To do so would require extensive measurements of the performance of the full-size and the model materials in the appropriate environments. Given such measurements, the acoustic design could proceed in a normal fashion without the aid of the model which would then effectively have served no purpose.

The real strength of the acoustic modelling technique lies not in the prediction of infinitesimal acoustic differences which are in any case unlikely to be reproduced exactly, but in the prediction of major acoustic details. As demonstrated in the work described in this report, mechanical and structural considerations can be used to predict the likely behaviour of materials and constructions to a reasonable degree of accuracy. It is the avoidance of gross acoustic defects and the consequent severe financial penalties that can justify the expense of constructing an acoustic model.

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*"thin" in this context means "of a thickness small in comparison with the other dimensions".

**"thick" in this context means "of a greater thickness than is usually considered in such cases".

Because it is only financially justifiable in those cases where remedial work is very expensive, it is unlikely that acoustic scale modelling would be applied to smaller rooms purely as an aid to room design. There are, however, other cases which might benefit from modelling. For example, the testing or development of acoustic materials or theories of sound propagation might economically be carried out at some scale other than the intended final size. As an example, the theoretical work described in Ref. 37 is currently being tested in the scale model reverberation room and will be reported in due course[38]. In these types of cases, the ease with which many changes may be made and the lower cost of smaller quantities of materials may be sufficient justification.

6. Subjective assessment

The completed model of Manchester Studio 7 was assessed subjectively by the usual method of recording in the model a selection of scaled musical passages which themselves contained no acoustical information about the room in which they were originally recorded[13]. The improved electro-acoustic transducers needed to do this are the subject of a companion report[15]. The recordings from the model were compared subjectively with recordings of the same musical passages recorded at normal speed in the full-size studio by means of monitoring loudspeakers and conventional microphones. The difficulties encountered in carrying out this comparison are properly the subject of the companion report.

Despite the differences in reverberation times between the studio and the model, the two recordings were very similar to each other. Some slight additional colorations from the scale model loudspeaker and a very slightly higher background noise level and non-linear distortion in the model recordings made two recordings instantly identifiable on an ABAB comparison. Otherwise, and especially acoustically, they were virtually identical.

7. Conclusions

Using mainly structural and mechanical considerations, a new acoustic scale model of Manchester, Studio 7 has been constructed. The model has reproduced the essential acoustic features of the studio to a reasonable degree of accuracy although the intermediate calculations of absorption coefficients show large differences. With the sole exception of the audience seating and its supporting structure, no corrections were made to the achieved acoustic performance of the model components in an effort to match their performance to that of the full-size components in the full-size studio.

Some of the value of the intermediate measurements of reverberation time in the full size studio during its construction has been lost because of the later changes in the acoustic treatment. These changes were brought about mainly by the excessive low-frequency absorption of the ventilation trunking which was not included in the earlier model because details of the trunking were not shown on the drawings at that time. Acoustic scale modelling can only be of value if comprehensive details of the proposed construction are available and remain unchanged from the end of the modelling. This does require that any features which could have acoustic implications are finally settled at a fairly early stage in the design.

Unless a great deal of time and effort and therefore expense is expended on getting exact and detailed matches between model and full-size components, the acoustic scale modelling technique is, at best, only as accurate as full scale acoustic design. Indeed, this is obvious – if it were more accurate then the full-scale design would be more accurate for the same reasons. The main advantage in modelling is that changes may be made much more readily. The final acoustic design of the model can be reproduced in the full-size structure only within the tolerances imposed by the reproduction of the model components at full-scale. These tolerances are almost inevitable so that it is equally inevitable that the full size structure will differ from the model in small details. It is these almost inevitable differences which made any effort to obtain exact matches a waste of time. It will generally be more economical to plan for at least one correction to the acoustic treatment in order to compensate for all of the small differences simultaneously. If the acoustic scale modelling has been carried out accurately using structural and mechanical considerations and all of the internal details have been considered then there will be no serious acoustic defects in the full size structure. The final correction will therefore be a relatively minor matter. It is in the avoidance of large errors and the consequential expensive, sometimes impossible remedial work that acoustic scale modelling has its real value as an aid to room design.

Other applications of acoustic scale modelling, for example, the development or testing of acoustic materials can be justified on other grounds.

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APPENDIX 1. THE VARIATION OF SCALED AIR ABSORPTION WITH MODEL SCALE.

The scaled air absorptions for different modelling scale factors over the full size equivalent frequency range of 1 to 10 kHz were calculated (from ref. 23). Table A.1 shows the results obtained. A relative humidity of 60% in the full size and 1% in the models has been assumed. Air absorption is given in nepers/m × 1000 and is for a temperature of 20°C.

TABLE A1. EFFECTIVE SCALED AIR ABSORPTION FOR A RANGE OF MODEL SCALE FACTORS.

Full scale		Model scale factor							
Freq. kHz	Air absorption (60% RH)	4	7	8	9	10	11	12	15
		Scaled air absorption at 1% R.H.							
1	0	1	1	1	1	1	1	1	1
2	1	1	1	2	2	2	2	2	2
3	3	2	3	3	3	4	4	4	5
4	5	3	5	5	6	6	7	7	9
5	8	4	7	8	9	10	10	11	14
6	11	6	10	11	12	14	15	16	20
8	19	10	17	19	22	24	26	29	36
10	30	16	26	30	34	37	41	45	56

The match between full-scale at 60% relative humidity and 8:1 scale model at 1% relative humidity can be seen to be very close. The errors at high frequencies which result from the use of other scaling factors are also evident. These errors cannot be corrected by the choice of other relative humidities, either in the model or in the full-size, because the effects of different humidities occur mainly at high frequencies in the full size and at the equivalent of low or medium frequencies in the scale model.

APPENDIX 2. DIMENSIONAL ANALYSIS OF PANEL RESONANT FREQUENCY

The study of vibrations in thin diaphragms has occupied many workers over a long period of time. As examples only, see refs. 31 to 36. Most of these are concerned with improved refinements of the basic theory and with the higher order modes. For the purposes of this elementary dimensional analysis, only the simplest theory and the fundamental mode will be considered.

a) For the inherent panel stiffness, Ref. 31 gives for the free vibration of a circular panel simply supported around the perimeter

$$f = 0.233t/R^2 \cdot \sqrt{\frac{E}{\rho(1-\sigma^2)}}$$

where f is the fundamental frequency
R is the radius
t is the thickness
E is Young's modulus
ρ is the density
and σ is Poisson's ratio

for a given material and proportions this simplifies dimensionally to

$$f \propto 1/L$$

where L is the generalized dimension of length.

b) For the panel vibrating as a limp mass with the compliance of the enclosed airspace.

$$\begin{aligned} \text{compliance of airspace} &= AD/\rho_0 c^2 \\ \text{acoustic mass of panel} &= \text{mass}/A^2 \\ &= \rho t/A. \end{aligned}$$

where A is the area of the panel.
D is the depth of the enclosed airspace
c is the velocity of sound in air
ρ_0 is the atmospheric air density

and other symbols are as previously defined. Thus, since

$$f = 1/2\pi\sqrt{\text{mass} \times \text{compliance}}$$

$$f = \frac{\rho_0 c^2}{D\rho t}$$

or, again, for the same material and proportions this simplifies dimensionally to

$$f \propto 1/L$$

c) the ventilation trunking, being highly assymetrical, (approximately, semi-infinite), can be treated as a wide beam spanning the short dimension. The dimensional analysis is similar to that in (a) above.

In each of the above cases, for the same material and geometric proportions, the resonant frequencies are inversely proportional to the scaling factor which is the required condition for scaling.

APPENDIX 3. ON THE USE OF ACRYLIC SHEET FOR THE MODULAR ABSORBER CONSTRUCTION.

A comprehensive analysis of the vibrations of the boxes forming the modular absorbers is exceptionally difficult because of the different and hard-to-quantify support conditions of the panel edges. In the low-frequency modules the sides and back of the boxes were firmly glued together whilst the front panel was loosely attached. This means that the front edge of the sides might approximate to being simply-supported but the remaining three edges would be more rigidly supported but by something which was itself not rigid. In fact, the box back and sides probably vibrates as a complex shell rather than an assembly of panels. No more than a very simplified dimensional analysis can be presented here in an effort to show how the resonant frequencies might depend on the material properties and dimensions. Fortunately, for the frequency range in which resonances of the side panels is important, the inherent panel compliance dominates the total compliance. The compliance of the enclosed airspace can therefore be neglected.

From Appendix 2, the resonant frequency of a simply supported panel was given as

$$f \propto t/R^2 \sqrt{\frac{E}{\sigma(1-\rho^2)}}$$

using the same terminology as Appendix 2.

For a given panel size

$$f \propto \sqrt{Et^2/\rho(1-\sigma^2)}$$

For two different materials and two different thicknesses

$$f_1/f_2 = \sqrt{\frac{E_1\rho_2(1-\sigma_2{}^2)}{E_2\rho_1(1-\sigma_1{}^2)}} \cdot t_1/t_2$$

If material 1 is "perspex" with a Young's modulus of 3×10^9, a relative density of 1.2 and a thickness of 1.5, material 2 is wood sheet with values of 10×10^9, 0.7 and 1.1 respectively* and the

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* All units have been omitted since they cancel out.

Poisson's ratios of the two materials are similar then

$$f_1/f_2 = 0.57$$

This indicates that, for the model size low frequency modular absorbers, the inherent fundamental resonant frequency of the "perspex" box is nearly an octave lower than it would have been if the box could have been constructed using properly scaled materials.

If the actual resonant frequency of the full size wooden panel is calculated using the equation

$$f = \frac{0.233t}{R^2}\sqrt{\frac{E}{\rho(1-\sigma^2)}} \qquad \text{from Appendix 2}$$

a value of 384 Hz is obtained. In fact, the full-size plywood panels were measured to have a resonant frequency of between 250 Hz and 315 Hz. The calculated and measured result are sufficiently close to each other to justify the validity of the equation, at least for the purposes of the dimensional analysis.